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Analytical modeling of a desiccant wheel

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ABSTRACT

A simple integral model is presented for a desiccant wheel. The original governing equations for a desiccant wheel were simplified to a set of linear ordinary differential equations and an analytical solution was obtained. A brief analysis is given about the solution regarding the non-dimensional numbers that decide the behavior of a desiccant wheel. From the solution, algebraic expressions were obtained for time-averaged heat and mass transfer rates and the results were compared with a numerical model and a set of experimental data in the literature. In comparison with the numerical model, relative error was found less than 12% at 120 °C regeneration temperature and 10% standard deviation was observed with the experimental data. The analytical model is considered capable of describing a symmetric desiccant wheel realistically.

Modélisation analytique d'une roue déshydratante

Mots clés : Modèle analytique ; Roue déshydratante ; Adsorption ; Déshumidification ; Régénération ; Rotatif

1. Introduction

Due to the increasing concerns on energy and environment, desiccant-assisted evaporative cooling (DEC) is receiving greater interest as an alternative to the conventional technology. In most solid DEC systems, desiccant wheels are used to transport moisture from one air stream to the other. Desiccant wheels have been popular research subjects, for which many studies are found in the literature. Most of them are, however, based on various numerical models (Ge et al., 2008) and analytical studies are scarce. Popularity of numerical approach is partly due to the complex nature of the problem. Although a numerical model, typically based on FDM (Finite Difference Method), may yield high accuracy, high complexity and excessive computing time are great

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Nomenclature

A	area, m^2
C_r^*	thermal time constant in Eq. (51)
C_p	specific heat, $J\ g^{-1}\ K^{-1}$
c	moisture concentration, $g\ m^{-3}$
D	mass diffusivity, $m^2\ s^{-1}$
h	air-side heat transfer coefficient, $W\ m^{-2}\ K^{-1}$
h_m	air-side mass transfer coefficient, $m\ s^{-1}$
i_{fg}	heat of adsorption, $J\ g^{-1}$
Ja	Jakob number defined in Eqs. (52) and (53)
k	thermal conductivity, $W\ m^{-1}\ K^{-1}$
L	channel length, m
Le	Lewis number
M	molar mass, $g\ mol^{-1}$
m	mass, g
Nu	Nusselt number
N	number of transfer units
$\dot{m}$	mass flow rate, $g\ s^{-1}$
$\dot{n}$	mass flux, $g\ m^{-2}\ s^{-1}$
P	perimeter, m
p	pressure, Pa
$\dot{Q}$	heat flow rate, W
$\dot{q}$	heat flux, $W\ m^{-2}$
R	universal gas constant, $J\ mol^{-1}\ K^{-1}$
Sh	Sherwood number
T	temperature, K
t	time, s
t^*	dimensionless time ($=t/\tau$)
u	mean velocity, $m\ s^{-1}$
u'	local velocity, $m\ s^{-1}$
$\dot{V}$	volume flow rate, $m^3\ s^{-1}$
w	water uptake (mass of adsorbed water per unit mass of dry desiccant)
x	coordinate in axial direction, m
y	coordinate perpendicular to the wall, m
z	coordinate along the perimeter, m

Greek symbols

α	thermal diffusivity, $m^2\ s^{-1}$
β	constant in Eq. (38)
Γ	mass flow rate per unit perimeter, $g\ m^{-1}\ s^{-1}$
γ	ratio of Jakob numbers in Eqs. (67) and (68)
δ	half thickness of desiccant wall or air channel, m
ε	macro-scale porosity
τ	duration of a process, s
ρ	density, $g\ m^{-3}$
Φ	non-dimensional index in Eqs. (88) and (89)
ϕ	relative humidity
η	dimensionless distance in Eq. (6)
θ	dimensionless heat flux in Eq. (46)
λ	constant in Eq. (56)
ν	constant in Eq. (57)
ψ	weighting factor in Eq. (25)
χ	humidity ratio
Ω	rotational speed, s^{-1}
ω	dimensionless mass flux in Eq. (47)

Superscripts

b	bulk or cross-sectional average
eq	equilibrium condition

Subscripts

0	initial condition
τ	final condition
1	dehumidification or present process
2	regeneration or previous process
anlt	analytical model
a	air
c	convection
DS	dry solid
i	inlet
m	mass transfer
num	numerical model
o	outlet
ref	reference condition
s	solid
t	heat transfer

disadvantages in many practical applications such as system optimization and seasonal simulation. Among the few analytical studies in the past, Banks (1985a,b) assumed that a desiccant wheel might be represented by the superposition of two heat-transfer regenerators driven by combined potentials and presented methods for predicting exit air conditions. Van den Bulck et al. (1985a,b) firstly assumed infinite transfer coefficients between air and desiccant in the wave analysis to develop an ideal dehumidifier model and later combined it with a FDM model to develop a semi-empirical effectiveness-NTU model. Both Banks (1985a,b) and Van den Bulck et al. (1985a,b) improved understanding of the heat and mass transfer in the desiccant dehumidifier by using the analogy to the well-established rotary heat exchanger analysis. However, they should eventually rely on intense numerical computation due to the nonlinear nature involved in the solutions, which restricts the practical applicability of their works to engineering practices. Lee et al. (2004) and Kim et al. (2011) presented analytical solutions of the simplified governing equations based on some ideal assumptions (linear temperature and concentration profiles in Lee et al. (2004); uniform heat and mass fluxes in Kim et al. (2011)) at the air-desiccant interface. In this study, it is attempted to develop a simple yet accurate integral model based on more accurate analysis of the heat and mass transfer process. Such a model would not only help understand the behavior of a desiccant wheel but also simplify much of the engineering practice in the field. In the following, it will be shown how an integral solution can be obtained from the governing equations and be used to develop an integral model. Then accuracy of the integral model will be assessed via comparison with a FDM model and a set of experimental data in the literature.